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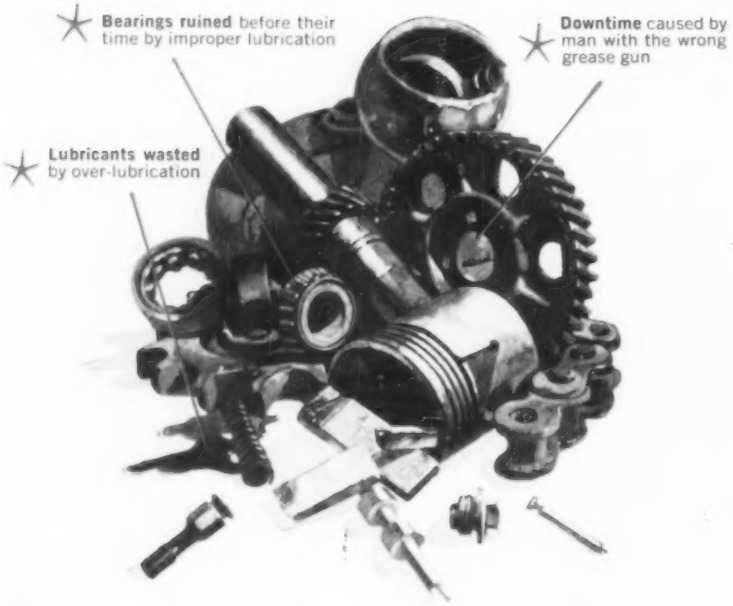
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A TECHNICAL PUBLICATION DEVOTED TO THE SELECTION AND USE OF LUBRICANTS

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VISCOSITY-II

THE JANUARY 1961 issue of this publication presented the first part of this article which discussed fundamental principles, and described several common methods and equipment used in measuring viscosity. This issue contains the second and concluding part which discusses: viscosity unit conversions; the effects of temperature, shear, pressure and nuclear radiation on viscosity; visco-elasticity; film lubrication; liquid flow in pipes and hydraulic systems.

VISCOSITY UNIT CONVERSIONS

A viscosity determined in a particular viscometer at a specific temperature can be converted to the equivalent viscosity which would have been obtained had the liquid been tested in some other viscometer *at the same temperature*. For the most accurate conversion, the determination temperature must be considered. Some of the wide variety of equations, tables, charts and nomographs which have been developed to facilitate such conversions are described in the following.

Kinematic to Saybolt Universal and Furol

ASTM Method D446-53 ("Standard Method for Conversion of Kinematic Viscosity to Saybolt Universal Viscosity"), ASTM Method D666-57 ("Standard Method for Conversion of Kinematic Viscosity to Saybolt Furol Viscosity") and ASTM Special Technical Publication No. 43B ("ASTM Viscosity Tables for Kinematic Viscosity Conversions and Viscosity Index Calculations") provide data in convenient form for these very common conversions.

Kinoshita[6] recently developed the following equation for converting kinematic viscosity ν (cs) determined at temperature t (°F.) into Saybolt Universal Seconds. The equation is regarded as highly accurate because it includes determination temperature and is satisfactory for kinematic viscosities greater than 2cs and temperatures between 70 and 300°F.

$$\text{SUS} = \frac{(4.6050 + 0.000297t)\nu}{1 - 10^{-0.07445\nu^{0.9538}}}$$

Kinematic to Redwood No. 1 (Standard)

The following Kinoshita[6] equation for this conversion is highly accurate for kinematic viscosities above 3 centistokes and temperatures between 70 and 210°F.

$$\text{Redwood Seconds} = \frac{(4.037 + 0.000344t)\nu}{1 - 10^{-0.06668\nu^{0.9704}}}$$

The following five groups of equations provide conversions in the opposite direction with sufficient accuracy for most purposes within their indicated ranges. In all cases "T" is the efflux time in seconds and "ν" the desired kinematic viscosity in centistokes.

Saybolt Universal to Kinematic

(T between 32 and 100 seconds) $\nu = 0.226T - \frac{195}{T}$

[6] Kinoshita, Masumi, J. Inst. of Pet., *43*, May 1957

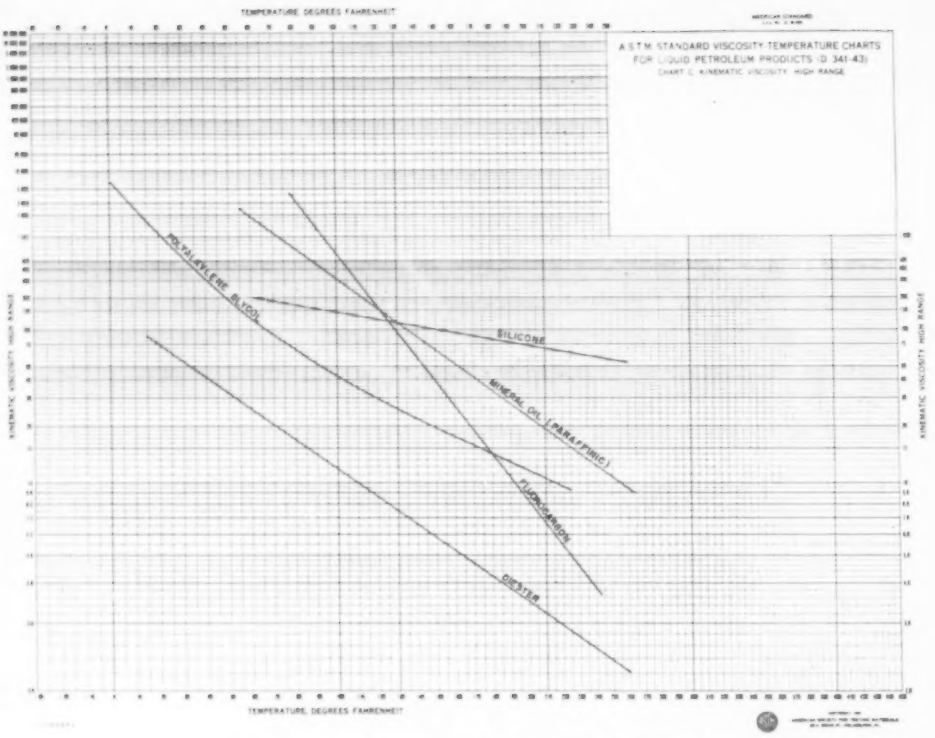

Figure 22 — ASTM viscosity-temperature chart with curves for several lubricants.

(T above 100 seconds) $\nu = 0.220T - \frac{135}{T}$

Saybolt Furol to Kinematic

(T between 25 and 40 seconds) $\nu = 2.24T - \frac{184}{T}$

(T above 40 seconds) $\nu = 2.16T - \frac{60}{T}$

Redwood No. 1 (Standard) to Kinematic

(T between 34 and 100 seconds) $\nu = 0.260T - \frac{179}{T}$

(T above 100 seconds) $\nu = 0.247T - \frac{50}{T}$

Redwood No. 2 (Admiralty) to Kinematic

(T between 32 and 90 seconds) $\nu = 2.46T - \frac{100}{T}$

(T above 90 seconds) $\nu = 2.45T$

Engler Degrees (E°) to Kinematic

(E° between 1.35 and 3.2) $\nu = 8.0E° - \frac{8.64}{E°}$

(E° above 3.2) $\nu = 7.6E° - \frac{4.0}{E°}$

A number of charts have been developed for the rapid approximate conversion of viscosity units, the most recent one being shown in Figure 27.

VISCOSITY-TEMPERATURE RELATIONSHIP

Viscosity is an oil's most important single property while temperature is the most important factor affecting viscosity. Viscosity decreases with increasing temperature but the *amount* of viscosity decrease depends both on the chemical nature of the oil and the magnitude of the temperature change. On the other hand many a mechanism is quite specific in demanding a particular lubricant viscosity at its *operating temperature* which latter rarely coincides with any of the few fixed standard temperatures that are customarily used in measuring viscosity. Thus the viscosity of a lubricant can be matched to the mechanism's requirements only if the lubricant's viscosity temperature characteristics are known over a considerable temperature range. The two most common methods for evaluating vis-

cosity temperature characteristics are the ASTM Charts (described in ASTM Method D-341-43, "Standard Viscosity Temperature Charts for Liquid Petroleum Products") and the Viscosity Index (V.I.) system of Dean and Davis.[7]

ASTM Charts

In 1942 these charts were modified slightly and adopted as ASTM Standards. Five varieties similar to that shown in Figure 22 are now available with various temperature ranges and either kinematic or Saybolt Universal viscosity scales. These charts facilitate simple graphical solutions to many viscosity-temperature problems. For example if viscosities of a given oil at two different temperatures (such as 100 and 210°F.) are plotted and a straight line drawn through them, then intermediate and neighboring viscosities at other temperatures may be read from the line. Extrapolations as high as the initial boiling point and as low as the cloud point of the oil are permissible: in fact the viscosities at 0°F. required by the SAE Crankcase Oil Viscosity Classification System (which may be as high as 48,000 Saybolt Universal Seconds) are habitually and most practically obtained by extrapolating the viscosity-temperature line established by their viscosities at 100° and 210°F. However a straight mineral oil changes to a non-Newtonian fluid at temperatures close to and below its cloud point, therefore viscosity determinations by extrapolation in this region are apt to be considerably lower than the apparent viscosities obtained by low shear viscometers such as capillary tubes. On the other hand, extrapolated viscosities agree more closely with apparent viscosities obtained under conditions of moderate or high rate of shear.

Since the line accurately portrays the viscosity-temperature characteristics of a particular oil, its *slope*, or angularity from the horizontal, is a measure of the oil's sensitivity to temperature changes. For example the viscosity temperature lines shown on Figure 22 for a mineral oil and several synthetic fluids illustrate that the particular silicone has the least sensitivity to temperature changes while the fluorocarbon has the most. Also of interest is the fact that while these viscosity-temperature charts were originally designed to give straight lines with *mineral oils*, three of the synthetics behave similarly and only the polyalkylene glycol deviates.

Viscosity Blending Chart

A number of formulas and charts have been developed in the past to either predict the required composition of a new two-component lubricating oil blend or to estimate the composition of an existing oil blend when the viscosities of the blend and of the two components are known. One chart commonly used today is shown in Figure 23 and is actually the ASTM Viscosity-Temperature Chart modified as follows.

The function log log $(\nu + 0.6)$, representing the vertical scale (ordinate) of the ASTM chart, is almost linearly related to the composition of a two-component oil blend as well as to the temperature variable, log $(t + 460)$. The ASTM vertical scale is accordingly used without change while that portion of the horizontal scale (abscissa) between 0° and 100°F. is simply re-labelled 0 to 100 per cent, and used to represent the per cent by volume of the high viscosity oil in the blend. To use the chart, plot the viscosity of the *low* viscosity oil on the zero per cent line and the viscosity of the high viscosity component on the 100 per cent line. If the two points are connected by a straight line, the required volumetric composition of any blend of intermediate viscosity may be read. Conversely, when the viscosities of the two components and of a blend made from them are known, the composition of the blend can be estimated.

Viscosity Index (VI)

Largely because of the convenience of the single simple number it uses to express the viscosity temperature characteristics of an oil, the Viscosity Index (VI) system devised many years ago by Dean and Davis continues to be widely used in the petroleum industry. This wholly empirical system is based upon comparison of viscosity measurements on fractions from crude oils "L" and "H" which were chosen because they seemed to possess the absolute maximum and minimum limits of viscosity temperature sensitivity and were accordingly assigned Viscosity Indices of 0 and 100 as the presumed end points of a 100 point Viscosity Index scale. While all other oils were expected to fall between these limits, the subsequent advent of solvent refining, the use of additives like the polymers, and the manufacture of synthetics have all produced lubricants that are far outside the Viscosity Index scale in both directions.

The Viscosity Index of an oil is obtained by determining its viscosities at 100°F. and 210°F., obtaining[8] the viscosities at 100°F. for those fractions of L and H that have the same viscosity at 210°F. as the unknown oil and substituting the three 100°F. viscosities in the following equation:

$$\text{Viscosity Index} = \frac{L-U}{L-H} \times 100 = \frac{L-U}{D} \times 100$$

where L = the viscosity at 100°F. of the zero VI fraction of oil L

H = the viscosity at 100°F. of the 100 VI fraction of oil H

U = the viscosity at 100°F. of the unknown oil

D = L—H

[7] Dean, E. W., and Davis, G. H. B., Chem. & Met. Eng. 36, 618 (1929)

[8] From ASTM Special Technical Publication No. 168 or No. 43B or ASTM Method D567-53 ("Calculating Viscosity Index")

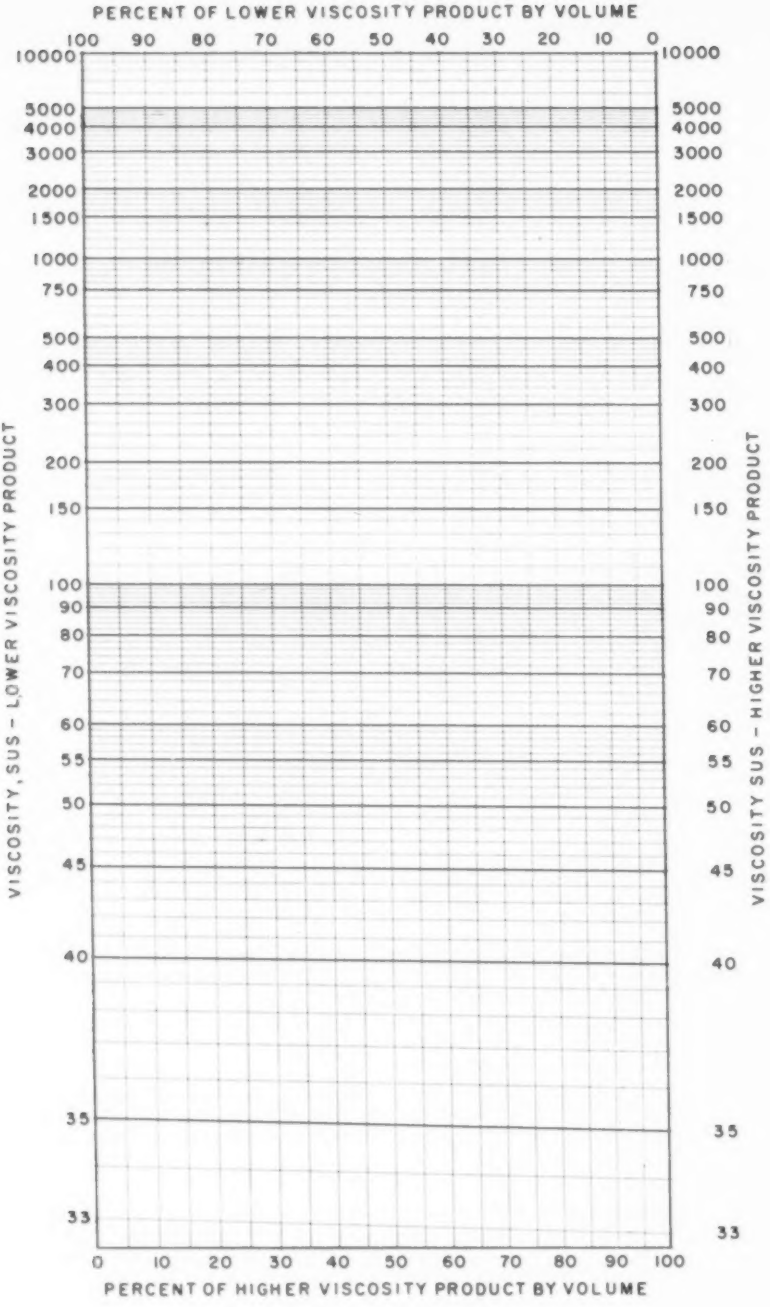

Figure 23 — Viscosity blending chart.

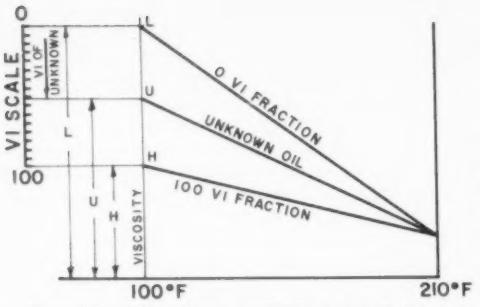

Figure 24 — Schematic representation of viscosity index.

Since the term L—H is a constant for any given 210°F. viscosity, some Viscosity Index tables perform the subtraction and list the result as equation term D.

Either Saybolt or Kinematic viscosities can be used, though the latter are preferred because of their greater accuracy. Figure 24 presents a schematic illustration of the Viscosity Index System.

Significance of Viscosity Index

As mentioned before, a prime requisite for any lubricant is that it have the proper viscosity under its actual operation conditions. If these involve wide variations in temperature, it is desirable from this viewpoint alone to use an oil having a high VI, i.e. one that shows relatively little change in viscosity with temperature. On the other hand a mechanism located in a heated room where only small changes in temperature are encountered is indifferent to viscosity index and other oil properties are more important. The only meaning that should be attached to VI is that it is an indication of the rate of change of viscosity with change in temperature—it tells nothing about other qualities of the oil. The application and range of temperatures that will be encountered are the only factors that should be considered in choosing an oil of high or low VI.

The Viscosity Index System has a number of limitations:

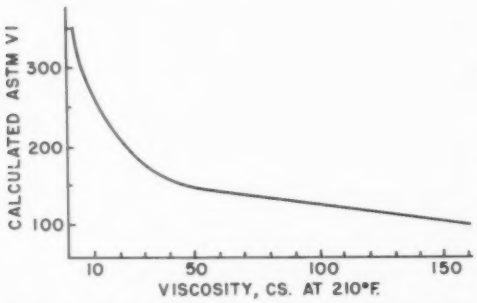

Figure 25 — Calculated viscosity index values for ideal oils (having no variation in viscosity with temperature) vs. viscosity at 210°F.

1. The system breaks down for very light oils having a viscosity below about 8 centistokes at 210°F.

2. As illustrated in Figure 25 the viscosity index of an oil having exceptionally good viscosity-temperature properties unfortunately depends on the oil's viscosity! The curve represents calculated VI numbers for hypothetical oils having *no* variation in viscosity with changes in temperature i.e. infinite viscosity index. The VI values actually vary from 100 to 350 depending on the viscosity of the oil at 210°F.

3. The VI system is based on arbitrary standards no longer suitable today.

4. Viscosity Index is not an additive property. The VI of an oil blend can not be computed reliably by taking the weighted average of the VI's of each individual component, particularly when widely different types of oil are involved.

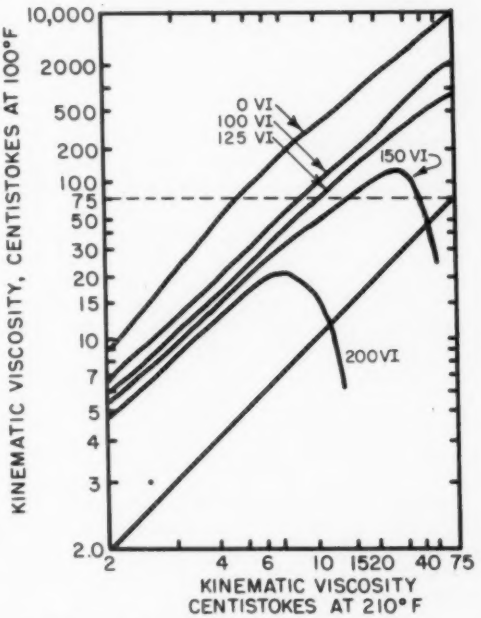

Figure 26 — Plot of lines of constant ASTM Viscosity Index.

5. In the range above 125 viscosity index, anomalous results are possible since two oils with the same viscosity at 100°F. but with different viscosities at 210°F. may have the same VI. This is shown in Figure 26 which is a plot of ASTM constant VI lines with viscosity scales on the same basis (the ordinate of the ASTM kinematic viscosity temperature chart). The lower diagonal line represents a series of hypothetical oils having the same viscosity at 100° and 210°F. All points above this

line mean that an oil decreases in viscosity as temperature is raised. From the 150 viscosity index line on this chart, it can be seen that two oils having a viscosity of say 75 cs at 100°F. and different viscosities at 210°F. (15 cs and 40 cs) can have the same VI of 150 which, of course, is not true. This latter irregularity has become more important since the introduction of polymer blended oils and synthetic oils of high VI.

Numerous methods and suggestions have been proposed to reduce or eliminate the shortcomings of the VI system or better yet to find a more fundamental and accurate means of expressing viscosity-temperature relationships. In 1956, Technical Committee B and Research Division VII of ASTM Committee D-2 appointed a special study group to investigate this problem. In an ASTM Bulletin[9] J. C. Geniesse outlined 9 different proposals to improve the present ASTM VI system and many more are available in the literature. Since the present VI system does have some value however and is firmly entrenched through several years of use, any changes will be only slowly adopted.

In 1959, Geniesse[10] reporting on the latest findings of the special study committee, stated that commercial interests wanted a simple comparative scale, retaining most of the present scale while scientific personnel want a more fundamental scale for research studies. To satisfy such divergent requirements, two systems will probably be required. The first of these would be a modification and improvement above 100 VI of the present ASTM system. The second system would adopt a more fundamental scale.

Viscosity Index Improvers

Mineral oils with viscosity indices above 100 can be made from a wide variety of crude oils by solvent refining, by selective blending of paraffinic base oils, by the addition of a relatively small amount of high molecular weight polymeric additives to base oils, or by combination of these methods.

Although polymer-blended oils with improved viscosity-temperature properties have been used increasingly only since the middle 1930's it is interesting to note that a United States patent was issued as early as 1867 for just such an oil. Eames and Seely with their "improved compound" consisting of rubber, a light hydrocarbon fraction and a "fixed" (fatty) oil claimed that it "increases the oil's fluidity at low temperatures, while at the usual summer heat the body of the oil is increased". This particular formulation, by the way, was not successful.

Modern polymeric VI improvers are usually composed of long-chain molecules having extremely high molecular weights between 5,000 and 20,000 and some as high as 700,000 have been used. Three principle types of compounds are butene polymers, polymethacrylates, and nitrogen-containing polymers, the latter also imparting dispersancy to an oil. These compounds have relatively low viscosity-temperature coefficients i.e. relatively flat viscosity-temperature slopes. When added to an oil they increase its viscosity relatively more at high than at low temperatures and thus make the viscosity-temperature curve of the blend more nearly horizontal.

Effectiveness of VI Improvers

The effectiveness of a polymeric type VI improver increases with increasing molecular weight of the polymer but is also dependent on the type of base oil used. The effectiveness can be evaluated by measuring and comparing the viscosity contributions of the polymer at 210° and at 100°F. Using kinematic viscosities measured on the blend and on the base oil itself at these two temperatures the *specific viscosities* at the two temperatures are calculated and their ratio determined as follows:

$$\nu_{sp} = \frac{\nu - \nu_o}{\nu_o}$$

where ν_{sp} is the specific viscosity, ν the viscosity of the blend and ν_o is the viscosity of the base oil.

$$\text{specific viscosity ratio} = \frac{\nu_{sp}(\text{at } 210°\text{F})}{\nu_{sp}(\text{at } 100°\text{F})}$$

When this ratio is greater than one, the polymer is a VI improver and the more it exceeds one, the better the VI improver. There are combinations of polymer and oil in which the viscosity contribution of the polymer is proportionally much the same at all temperatures. Such polymers merely thicken the oil without affecting its viscosity temperature characteristics, and will have a specific viscosity ratio of less than one.

Mechanism of VI Improver Action

Reverting to Figure 1, the viscosity of an oil can be thought of as the internal fluid friction due to the oil molecules moving past each other. If this movement is hindered in any way, a higher viscosity will result. Lowering oil temperature is one method of accomplishing this. Another way to restrain the oil molecules' movement and increase viscosity is to add some extremely large molecules like those of a high molecular weight polymer having 350 to 1400 or more carbon atoms. The more of these polymeric molecules that are added or the larger their size, the higher the resultant oil viscosity will be.

The molecules of a VI improver in an oil solvent have the property of increasing in size with increasing temperature and hence contribute more to raising viscosity at the higher than at lower temperatures.

[9] ASTM Bulletin No. 215, July 1956.
[10] 1959 Fifth World Petroleum Congress, Section V, Paper 30A, "A Study of the ASTM VI Problems".

The ability of a polymer molecule to change in size with temperature is dependent on its solubility in the solvent. In a *poor* solvent, the internal attracting forces and the intermolecular forces cause the long chain molecules to curl up into small compact molecules attracted to each other with a resultant average molecular configuration represented by a tightly contracted mass. In this compact state, the polymer contributes relatively little to increasing viscosity. With a rise in temperature, however, the molecules increase in size, thus contributing more to increasing the viscosity. This nullifies somewhat the normal effect of the higher temperature in reducing the oil's viscosity. All told, the thickening action of a polymer in a poor solvent is greater at higher temperatures with the result that viscosity-temperature characterisics are improved.

In a *good* solvent the polymer molecules, because of low intra and intermolecular forces, are uncurled with a resultant greater volume. Hence, with increasing temperature the polymer, being already enlarged, hardly contributes to raising viscosity.

From the above supposition, it is seen that the effectiveness of a VI improver is dependent on its solubility characteristics in an oil. The best VI improvers are linear polymers that are near their precipitation point at the lowest temperature encountered in service but are completly uncurled or enlarged at the maximum temperature encountered.

EFFECT OF SHEAR ON VISCOSITY

Since a polymer-thickened oil is non-Newtonian in character, its viscosity will vary with rate of shear as shown in Figure 5. However, another important aspect of rate of shear is that when oils containing VI improvers are subjected to high rates of shear, losses in viscosity and in VI are often encountered. The extent of these losses is greater with higher shear rates and with higher molecular weights. Thus the choice of a VI improver is often a compromise between maximum effectiveness and shear (viscosity) stability.

The loss in viscosity and VI under high shear rates can be either *permanent* (due to a mechanical or chemical breakdown of the large polymer molecules into smaller molecules) or it can be *temporary* (due to orientation of the long-chain polymer molecules in the direction of flow thus offering less resistance to flow resulting in less viscosity). When shearing stress is removed in the "temporary" cases, the oils regain their original viscosity.

Fleming and others[11] conducted tests on automobile engines using various combinations of base oils and VI improvers. They found permanent losses in viscosity at 210°F. ranging from 0.8 to 4.5 Saybolt seconds and losses in VI from 3 to 9 units which are not excessive. They also measured viscosity of a polymer oil at varying shear rates and found only a 6 per cent temporary loss in viscosity at shear rates of 500,000 to 700,000 sec.$^{-1}$

The American Petroleum Institute (API) in conjunction with the ASTM conducted an investigation on viscosity variation with shear rates up to one million reciprocal seconds which is reported in ASTM Special Technical Publication No. 111, "Symposium on Methods of Measuring Viscosity at High Rates of Shear". Two of the oils studied, designated API 103 and 104, with a viscosity of about 155 SUS at 100°F., were polymer-blended oils. The work indicated that these low viscosity oils undergo a 30-40 per cent temporary viscosity loss at shear rates of 50,000 to 1,000,000 sec.$^{-1}$ These shear rates are comparable to those found in automobile engines. Oil API-104 which contained a higher molecular weight polymer than oil API-103, showed a greater temporary loss in viscosity.

Klaus and Fenske[12] recently reported on viscosity-shear characteristics of lubricants and related results to typical parts of hydraulic systems and other lubricated mechanisms. In measuring shear severity of hydraulic pumps, they found a permanent viscosity decrease of 2 to 21 per cent on a particular polymer-blended hydraulic oil. They concluded that in typical service permanent viscosity loss can be kept low and insignificant if the oil and polymer are properly selected and blended.

Although considerable work has been done in studying the effects of high shear rates on the viscosity of polymer-blended oils, more studies are needed in this very important field because of the numerous variables and different types of oils and polymers involved.

EFFECT OF PRESSURE ON VISCOSITY

When lubricating oils are subjected to high pressures, of the order of several thousand psi (pounds per square inch), their viscosity increases markedly. When pressure on an oil film lubricating a bearing reaches this order of magnitude, its effect on viscosity should be considered. Only within the past 20 years however has the effect of pressure on viscosity relating to bearing lubrication become significant. Before, this effect usually was not considered in bearing theory since the pressures encountered were so moderate that viscosity was not materially affected. Today this is no longer true. Lubricant film pressures up to 200,000 psi and more may be encountered in ball and roller bearings and in gears. In this range, pressure affects viscosity very much and must be considered in the lubrication and design of these elements.

In journal-bearing calculations, pressure effects are usually disregarded on the assumption that the

[11] Fleming, C. L., Geddes, B. W., Hakala, N. V. and Weisel, C. A. S.A.E. Quart. Trans. 4, 410 (1950)

[12] Klaus, E. E., and Fenske, M. R.,—Lubrication Eng., Vol. 11, p. 101, March-April 1955.

VISCOSITY CONVERSION CHART

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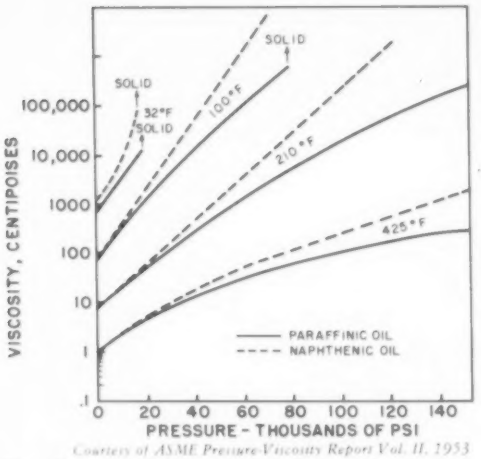
LINE UP STRAIGHT EDGE SO CENTISTOKE VALUE ON BOTH KINEMATIC SCALES IS THE SAME. VISCOSITIES AT THE SAME TEMPERATURE ON ALL SCALES ARE THEN EQUIVALENT.

TO EXTEND RANGE OF ONLY THE KINEMATIC, SAYBOLT UNIVERSAL, REDWOOD NO. I AND ENGLER SCALES: MULTIPLY BY 10 THE VISCOSITIES ON THESE SCALES BETWEEN 100 AND 1000 CENTISTOKES ON THE KINEMATIC SCALE AND THE CORRESPONDING VISCOSITIES ON THE OTHER 3 SCALES. FOR FURTHER EXTENSION, MULTIPLY THESE SCALES AS ABOVE BY 100 OR A HIGHER POWER OF 10.

(EXAMPLE: 1500 CENTISTOKES = 150 X 10 CS ≐ 695 X 10 SUS = 6950 SUS)

Figure 27 — Nomograph for conversion of viscosity units.



Courtesy of ASME Pressure-Viscosity Report Vol. II, 1953

Figure 28 — Viscosity-pressure curves for paraffinic and naphthenic oils at several temperatures.

viscosity increase due to pressure is offset by a decrease in viscosity due to higher operating temperatures. For example, average figures from considerable data indicate that a pressure increase of about 500 psi has the equivalent but opposite effect on viscosity as a temperature increase of 2°F. However, even here pressure effects may play an important part since pressures in the range of 5,000 to 10,000 psi are sometimes encountered in journal and thrust bearings. At 10,000 psi, viscosities are 4 to 10 times those at atmospheric pressure.

In the past few years an increasing amount of work has been done on viscosity-pressure properties, notably by the American Society of Mechanical Engineers (ASME)[13]. Hersey and Hopkins' report,[14] which analyzes and coordinates data from 12 major investigations, gives a bibliography on this subject covering about 200 references from 1892 to 1952.

A number of empirical equations have been derived relating viscosity to pressure. The following logarithmic one is most commonly used, although valid only over a moderate pressure range:

$$Z_p = Z_a e^{kp}$$

where

Z_p = absolute viscosity at pressure p.
Z_a = absolute viscosity at atmospheric pressure
e = 2.71828
k = a constant for a given oil and temperature.
p = pressure, psi.

Similar to the effect of temperature, pressure has less effect on the viscosity of paraffinic than on naphthenic oils. The viscosities of "fixed oils", (animal and vegetable) vary the least with pressure. Figure 28 gives viscosity-pressure curves[13] for paraffinic and naphthenic oils at several temperatures. Viscosity increases with pressure at constant temperature however the rate of increase is less at the higher temperatures. In addition, the more paraffinic oils solidify at certain critical pressures depending on the temperature while the other oils steadily increase in viscosity. If temperature is low enough, naphthenic oils will also solidify.

Roughly, most petroleum products at room temperature increase in viscosity by a factor of about 2.3 when the pressure is increased to 5000 psi. Other general correlations between pressure and viscosity of petroleum oils are:

1. The same increase in pressure will have a greater effect on viscosity at high pressures than at low pressures.
2. Oils of lower viscosity are less affected by pressure than oils of higher viscosity.
3. An increase in pressure will increase the viscosity index of an oil: naphthenic oils show a greater relative increase than paraffinics.

The relationship between VI and increase in pressure is shown in the tables below.

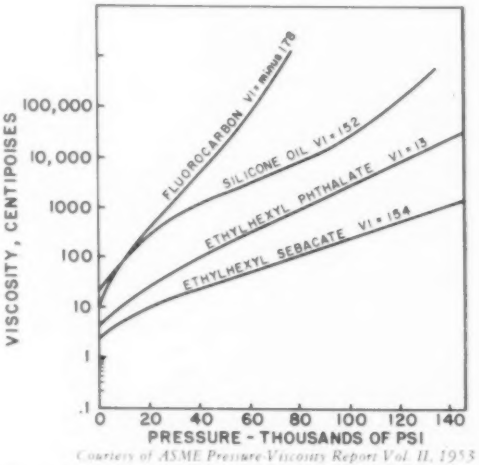
PARAFFINIC OIL

Pressure, psi	Viscosity, cs. at 100°F.	Viscosity, cs. at 210°F.	VI
Atmospheric	52.5	6.80	90
20,000	810	43.5	100
40,000	8700	195	125

NAPHTHENIC OIL

Pressure, psi	Viscosity, cs. at 100°F.	Viscosity, cs. at 210°F.	VI
Atmospheric	55.4	5.80	16
20,000	2186	53.5	54
40,000	91,000	454	115

Figure 29 shows the variation of viscosity with pressure at 210°F. for several synthetic oils. Note



Courtesy of ASME Pressure-Viscosity Report Vol. II, 1953

Figure 29 — Variation of viscosity with pressure for several synthetic oils at 210°F.

[13] Pressure-Viscosity Report, Vol. I-II, ASME, 1953.
[14] Hersey, M. D., and Hopkins, R. F., "Viscosity of Lubricants Under Pressure", ASME, 1954.

that the phthalate and the sebacate oils have somewhat similar changes in viscosity with pressure but a large difference in VI.

Although a considerable amount of data relating viscosity to pressure for a number of different oils has been collected, there still is a scarcity of data correlating viscosity-pressure effects with actual lubrication performance on a variety of machine elements. In addition, accurate and fundamental relationships for viscosity-pressure changes need to be developed. In connection with an ASME project some recent contributions to the knowledge in this area have been made. Needs[15] reported on the viscosity-pressure effect in a journal bearing. Hersey and Staples[15] described experiments on imperfect lubrication and viscosity-pressure effects and Sternlicht[15] reported on the influence of pressure and temperature on oil viscosity in thrust bearings. More data of this type are needed for a still better understanding of pressure effects as related to lubrication.

Since the viscosity of an oil depends on both pressure and temperature, it would be highly desirable to have an accurate and fundamental expression of viscosity-temperature-pressure relations applicable to all lubricants. Work of this nature has been done and a number of equations have been derived but they all have limitations and more studies are needed.

Consideration of viscosity-pressure effects has resulted in the desirable use of lower viscosity oils than the "classical" bearing equations would have permitted. These oils result in less fluid friction, lower operating temperatures and lower power losses. As more data of this type are collected an even more intelligent choice of lubricants for a variety of applications will be possible.

EFFECT OF NUCLEAR RADIATION ON VISCOSITY

With the advent of the "Atomic Age" or more correctly the "NEA" (Nuclear Energy Age) in 1942, users and suppliers of petroleum products have had another factor to contend with regarding viscosity (and other properties for that matter) of lubricants and fuels, namely, the effects of nuclear radiation on viscosity. Present data show that whenever the above materials are subjected to high dosages of radiation in nuclear reactors,[16] serious degradations sometimes take place which affect their performance. Work is being done not only in studying changes in viscosity with irradiation but also in developing new radiation-resistant materials.

Hydrocarbon oils exposed to high energy radiation increase in viscosity, the extent depending on their composition, radiation dosage and radiation type. In general, paraffinic oils show a greater increase in viscosity than naphthenic oils. Synthetic lubricating oils usually show an even greater increase in viscosity than the paraffinic ones. Viscosity increases at 100°F. vary from only a few per cent to 1000 per cent and in extreme cases the oil becomes a solid.

Carroll and others[17] in studying the radiation stability of hydrocarbon fuels (gasoline, kerosine, JP-3, JP-4, JP-5 jet fuels) found all fuels increased in viscosity upon irradiation but only significantly at dosages above 10^9 roentgens where some increased from one to 200 centistokes at 100°F.

Oils containing VI polymers behave in an unusual manner when irradiated. Initially at relatively low dosages, viscosity decreases. Then as radiation is increased still further, viscosity increases to even several times the original value. The over-all effect on viscosity depends on several factors. These include molecular weight and concentration of the VI improver, radiation type and dosage level and the use of inhibitors. Certain compounds, for example didodecyl selenide, have been found very effective in reducing radiation damage as indicated by viscosity changes.

Rice[18] in studying the gamma radiation stability of polymer-blended hydraulic oils found that increasing radiation exposure results in decreases in viscosity index.

Mechanism of Radiation Effects

Viscosity increases upon irradiation are largely due to a process termed crosslinking which is similar to polymerization. Molecular fragments formed in an oil during irradiation may also recombine to form molecules larger than the original oil molecules. Lighter molecules and gases such as hydrogen and methane, may also be formed.

When polymers are irradiated, broken bonds result and the cleavage is permanent when the break occurs in the main chain of the polymer. This causes a lowering of viscosity at initial relatively low dosages. However, as the radiation level increases, crosslinking predominates with a resultant rise in viscosity.

With an ever increasing use of nuclear energy, effects of nuclear radiation on viscosity will become more widespread and will have to be considered, (where dosages are high enough) before a proper choice of lubricants can be made for specific applications.

VISCOELASTICITY

In the preceding sections several factors affecting the viscosity of lubricants have been discussed, mainly temperature, pressure and shearing effects.

In recent years, other influences on viscosity have

[15] ASME Transactions, Vol. 80, p. 1099-1112, 1958.
[16] "Nuclear Power Plants", Magazine Lubrication, Sept. 1959
[17] Carroll, J. G., Bolt, R. O., Bert, J. A., ASTM Special Technical Publication No. 220, p. 108, 1958
[18] Rice, W. L. R., WADC Technical Report 57-573, Dec. 1957

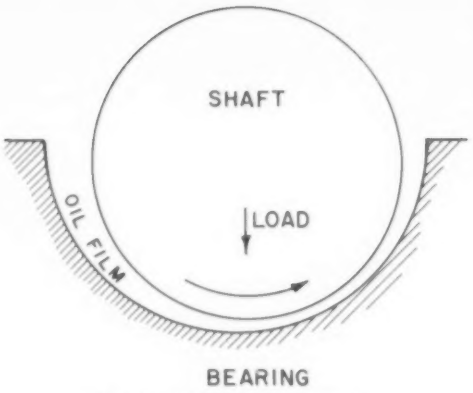


Figure 30 — Oil film wedge in bearing.

been assumed including presence of polar compounds, electric fields and viscoelasticity to mention a few. These were postulated in an attempt to explain certain anomalies in the lubrication of highly-loaded high-speed gears and bearings, for example, decreasing load-carrying capacity with increasing speed of spur gears up to a critical speed, and then increased load-carrying capacity with further increasing speed. Of these possible influences, *viscoelasticity* appears to be the most important part of the explanation for such unusual lubricant behavior.

Viscoelastic behavior of a lubricant is exhibited when the lubricant is subjected to rapidly changing stresses — for example, high pressures and shear rates occurring in non-steady states. When these occur, the lubricant no longer acts solely as a viscous flowing liquid, but momentarily assumes some characteristics of an elastic solid. Time is very important in this phenomenon since viscosity is dependent on both the stress applied and the rate of change of stress. Times involved range from a few microseconds to several minutes.

When a lubricant is subjected to a high pressure it is suspected that initially (possibly in the first few microseconds) a greater viscosity will be obtained than that of a steady-state measurement at the same pressure. This transient effect will disappear with time and the viscosity should decrease to the steady-state value. The duration of this transient effect is known as the "relaxation time" which is also important in the study of viscoelastic effects.

Unfortunately, very little is presently known about viscoelastic properties that could be used by lubricant suppliers and designers of machine elements. However, in the past few years, a great deal of interest has been shown in this behavior of lubricants and more data and a better understanding of lubrication processes will be forthcoming. The ASME Lubrication Division, to mention one group, has established a special study group to consider this problem and to recommend research programs for collecting basic data.

FILM LUBRICATION

The primary function of any lubricant is to reduce friction between elements of a mechanism moving relative to each other. The three types of fluid lubrication are:

1. Hydrodynamic (also known as "complete", "thick film" or "fluid film" lubrication).
2. Boundary or thin film lubrication.
3. Incomplete (also known as "semifluid" or "quasihydrodynamic" lubrication).

The type of lubrication depends primarily upon the thickness of the lubricating film, which in turn depends on load, time, speed, bearing design, temperature and actual lubricant viscosity in the bearing.

Hydrodynamic Lubrication

In this type, the moving parts are separated from each other by a film thicker than about $\frac{1}{40{,}000}$ of an inch. Almost all bearings depend for their operation upon oil-film pressures which carry the load. This film pressure is generated by hydrodynamic action as illustrated in Figure 30. The shaft or journal has a diameter slightly less than the bearing. As the shaft rotates, its center is displaced from the bearing center and a wedge-shaped passage is formed between the two surfaces. As oil is dragged into this passage, pressure is developed due to the viscous nature of the oil and the shaft is supported on the fluid film. Viscosity of the oil is the important factor in this type of lubrication. Under ideal hydrodynamic action, there is no wear

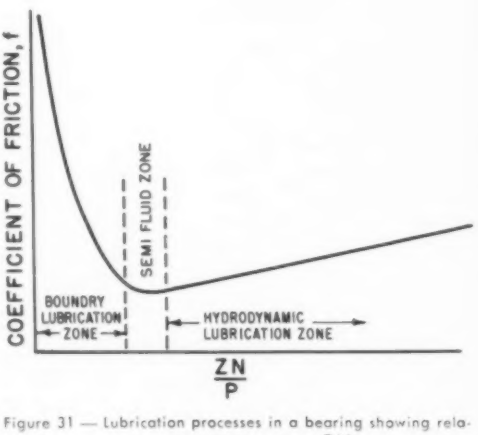


Figure 31 — Lubrication processes in a bearing showing relationship between $\frac{ZN}{P}$ and coefficient of friction.

since the moving parts do not touch each other and only low friction losses are encountered which are due solely to the viscosity of the oil. Coefficients of friction amount to only a few hundredths or thousandths. For these reasons, whenever possible, mechanisms are designed to operate in the region of hydrodynamic lubrication.

In some hydrostatic thrust bearings[19] the load is initially raised and subsequently carried by oil pressure externally supplied by a pump. As a result, the lubricating oil always and completely separates the bearing surfaces.

When a fluid film has been established between a rotating shaft and its bearing, the coefficient of friction depends on the speed of rotation, the load on the bearing, the oil viscosity at the bearing temperature and bearing design and condition. For given bearing dimensions, *and hydrodynamic lubrication*, the coefficient of friction, *f*, is an *approximately linear function* of the dimensionless parameter:

$$\frac{ZN}{P}$$

where Z is the absolute viscosity (centipoises), N is the speed, (revolutions per minute), and P is the load (pounds per square inch on the projected area of the bearing). This relationship is shown in Figure 31.

As the number $\frac{ZN}{P}$ is reduced (e.g. by increasing the load on an oil film, by decreasing speed of rotation, or by lowering viscosity) the oil film decreases in thickness to such a point that the laws of hydrodynamic lubrication no longer apply. Referring to Figure 31, it is seen that the coefficient of friction goes through a minimum point, this region being called the semi-fluid zone where both viscosity and surface or boundary forces play a role in lubrication. At still lower values of $\frac{ZN}{P}$ the coefficient of friction rises sharply. In this zone, where boundary lubrication occurs, the oil film has broken and there is contact between the rubbing parts, hence the physical and chemical properties of the rubbing solids are now important. Protection against the destructive effects of boundary lubrication is provided by additives which furnish either a very thin lubricating film of *physically adsorbed* molecules or which react chemically with the metal surfaces to form a thin anti-weld coating.

FLOW IN PIPES

Pipes play a very important role in transporting fluids conveniently and economically. In the United States alone there are already about 220,000 miles of pipelines (about as much as of railroad mileage) and more pipelines are being built.

[18] "Stationary Turbo Generators", Magazine Lubrication, July 1960

Fluid flow in pipes is dependent on several properties of the fluid (e.g. viscosity, density) and of the pipe system (length, pressure or head, flow velocity, diameter, shape, and pipe roughness). In any completely filled pipe through which a fluid is flowing, friction causes a continuous loss of head or "pressure drop" which must be determined when designing the pipe and in many other engineering problems which involve fluid flow.

The fundamental equation for calculating the loss of head in circular pipes (generally known as the D'Arcy or Fanning formula) is:

$$h = f \frac{L}{D} \frac{V^2}{2g}$$

where h is the loss of head (feet, equivalent to height of a column of liquid giving the pressure necessary to overcome the friction loss)
f is the friction factor (a dimensionless quantity)
D is the internal diameter (feet)
L is the length of pipe (feet)
V is the velocity of flow (feet per second)
g is the acceleration of gravity (32.2 feet per second per second)

In flow problems, L, D and V usually are known or given, thus if the friction factor, f, can be determined, then the loss of head can be calculated. This is where viscosity plays its part as an important component of the previously-defined Reynolds Number (R) which for this purpose is expressed in English system units as follows:

$$R = \frac{DV\rho}{\eta} = \frac{DV}{\nu}$$

in which D is the pipe diameter (feet), V is the flow velocity (feet per second), ρ is the fluid density (pounds per cubic foot), η is the absolute viscosity of the fluid (pounds of mass per foot second) and ν is the kinematic viscosity (square feet per second: one stoke is equivalent to 0.001076 square feet per second).

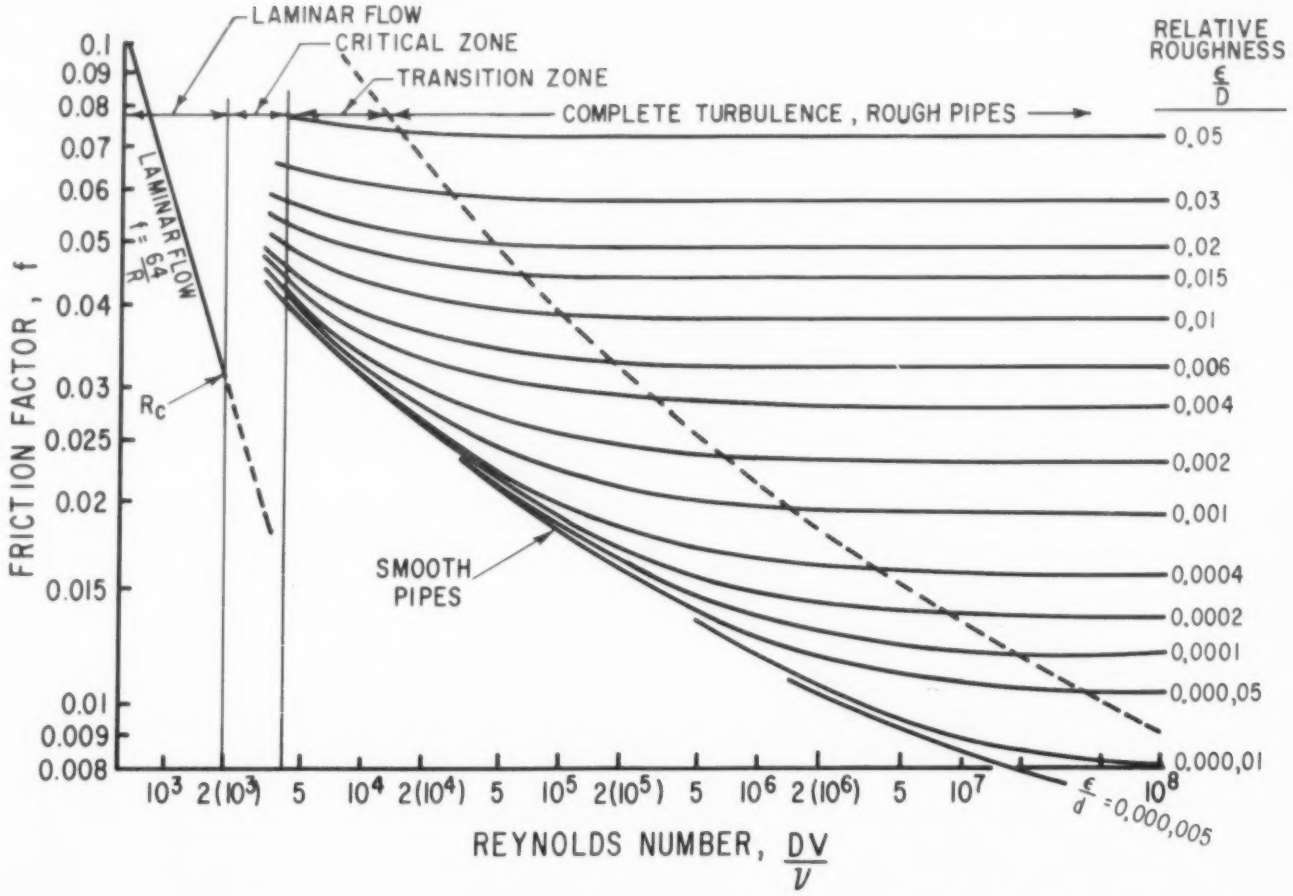
Laminar Flow

In laminar or streamline flow (Reynolds Number less than 2,000) the friction factor is inversely proportional to R and is given by the equation $f = \frac{64}{R}$ and is independent of the roughness of the pipe. This is shown graphically in the laminar flow zone of the diagram presented in Figure 32, which was developed by Moody.[20]

Turbulent Flow

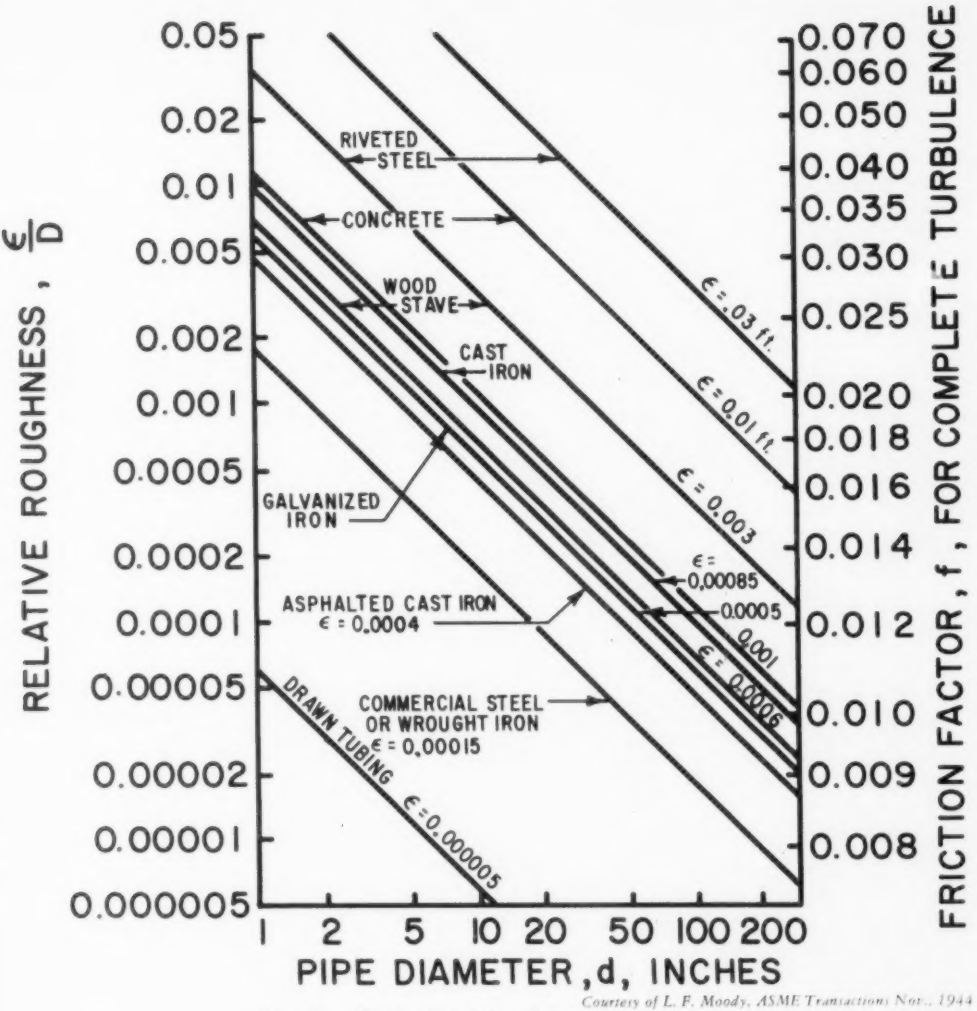
Between Reynolds Numbers of about 2,000 and 4,000 labelled "critical zone" in Figure 32, flow behavior may be either laminar or partially turbu-

[20] Moody, L. F., ASME Trans. Vol. 66 No. 8 p. 671 Nov., 1944.



Courtesy of L. F. Moody, ASME Transactions Nov., 1944

Figure 32 — Friction coefficient vs. Reynolds Number for flow in circular pipes.



Courtesy of L. F. Moody, ASME Transactions Nov., 1944

Figure 33 — Chart for determining relative roughness of pipes.

lent and is not a definite function of Reynold's number. Beyond 4000, flow again takes on a definite trend and there are two distinct regions, the region of transition where flow is partially laminar and partially turbulent and secondly, the complete turbulence region, sometimes called the "rough pipes" region.

Turbulent flow in a pipe system should be avoided whenever possible because it causes excessive pressure drop, more power is consumed in forcing the liquid through the lines and more heat is developed in the system.

Relative Roughness

In the transition and complete turbulence zones, the friction factor is dependent on the roughness of the pipe's surface and the diameter of the pipe. This parameter is called "relative roughness" and is defined as $\frac{\epsilon}{D}$ where ϵ, called the absolute roughness, is the average length in feet of the roughness protrusions on the inside wall surface and D is the inside pipe diameter in feet. Moody[20] also developed a chart illustrated in Figure 33 which evaluates the relative roughness of a number of commercial pipes in a wide range of diameters. Successive use of Figures 33 and 32 (in that order) permit a reasonable evaluation of the applicable friction factor and calculation of the loss of head.

Reynolds Number Above 4,000

In the transition zone, where most of the flow conditions in practice are found, the friction factor is a function of both Reynolds Number and relative roughness hence viscosity plays an important part.

In the complete turbulence region, the friction factor becomes constant for a specific relative roughness and is no longer a function of Reynolds Number: in other words, the friction factor is dependent upon relative roughness alone.

HYDRAULIC SYSTEMS

An hydraulic system involves essentially the generation, transmission and accurate control of power by fluid methods. The "life line" of any hydraulic system is the piping or tubing circuits which carry the hydraulic fluid under pressure. Since the flow of a liquid through pipes is involved, the foregoing discussion applies equally well here.

As in the case of an oil lubricant, the most important single property of an hydraulic oil is also its viscosity. Three important factors are:

1. Viscosity must be compatible with the mechanical clearances and fits of the various elements in the hydraulic system: to perform efficiently the oil must flow readily.
2. In systems subjected to a wide temperature range, changes in oil viscosity with changes in temperature may be minimized by using an oil of high viscosity index.
3. Changes in viscosity with change in pressure and rate of shear should be small.

Factors that are affected by the viscosity of an oil and which preclude 100 per cent efficient operation of an hydraulic system include oil leakage, particularly in the pump, cavitation (formation of pockets or cavities in a liquid stream), mechanical friction and rate of response.

From the viscosity standpoint the selection of an oil is usually a compromise since it is governed by operating temperatures, pressures, design features, size and condition of the various elements in the system. Generally, the viscosity of an hydraulic oil in service should not go above 4000 Saybolt Universal seconds (SUS) nor below 45 SUS.

VISCOSITY AS APPLIED TO ROCKET PROPULSION

For the next decade or two and probably even longer, liquid and solid propellants will be most practical and commonly used sources of energy for the propulsion of rockets in ballistic missiles and satellite and space vehicles. Although in these applications viscosity is not the most important single property of the propellant, it nevertheless plays an important role.

Regarding solid propellants, viscosity is used as a control in the complex manufacturing steps. Physical properties of certain solid propellants are reflections of the visco-elastic behavior of some of the ingredients.

With liquid propellants, viscosity plays an even more important role in that performance of the propellant and design of the "hardware" (pumps etc.) of the rocket engine are governed to some extent by viscosity. A liquid propellant must be sufficiently fluid so that it can be pumped easily at the temperature of launching. An upper limit of about 20 centipoises has been set for the viscosity of liquid propellants. In practice, however, materials being considered for use have a viscosity closer to one centipoise. In addition, liquids should have a small change in viscosity with temperature to decrease power requirements for pumps, and to minimize changes in bi-propellant mixture ratios as temperature changes. Variable mixture ratios result in inefficient utilization of the available propellants.

The heat transfer characteristics of liquid propellants which are also used as rocket motor coolants increase with decreasing viscosity.

SUMMARY

Viscosity is the most important single property of a lubricant. This article has covered several aspects of this broad subject such as fundamentals; measurement and instruments; units and conversions; effects of temperature, pressure, rate of shear, and nuclear radiation on viscosity; VI improvers; and viscoelastic properties. The significance of viscosity in relation to lubrication and flow in pipes has been discussed.

Although considerable data have been collected in this field during the past decade or two, even more are needed for a better understanding of the lubrication process and to aid in developing lubricants to meet the ever increasing demands being imposed on them.

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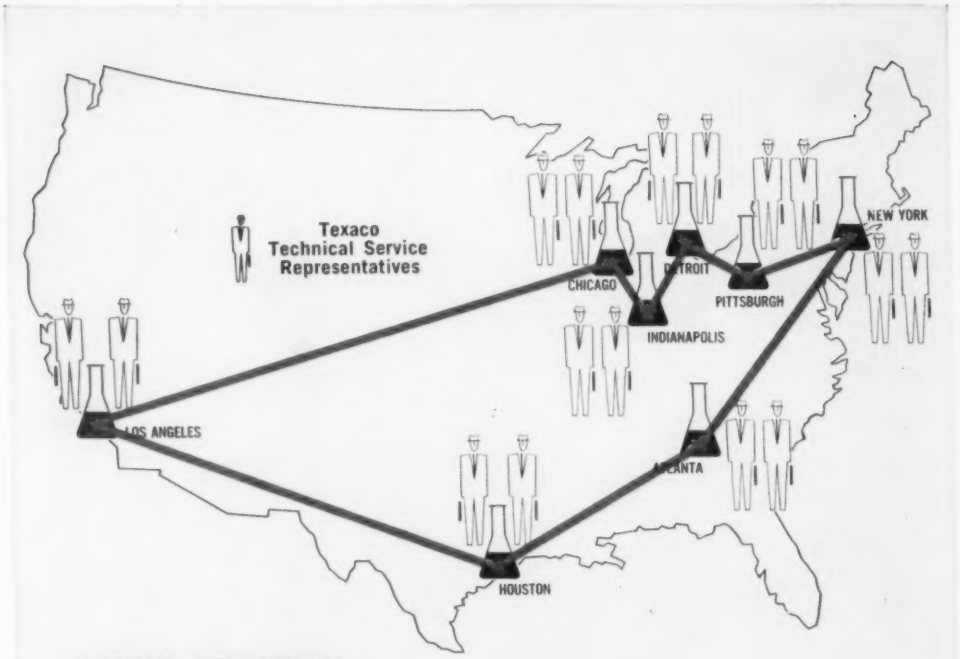

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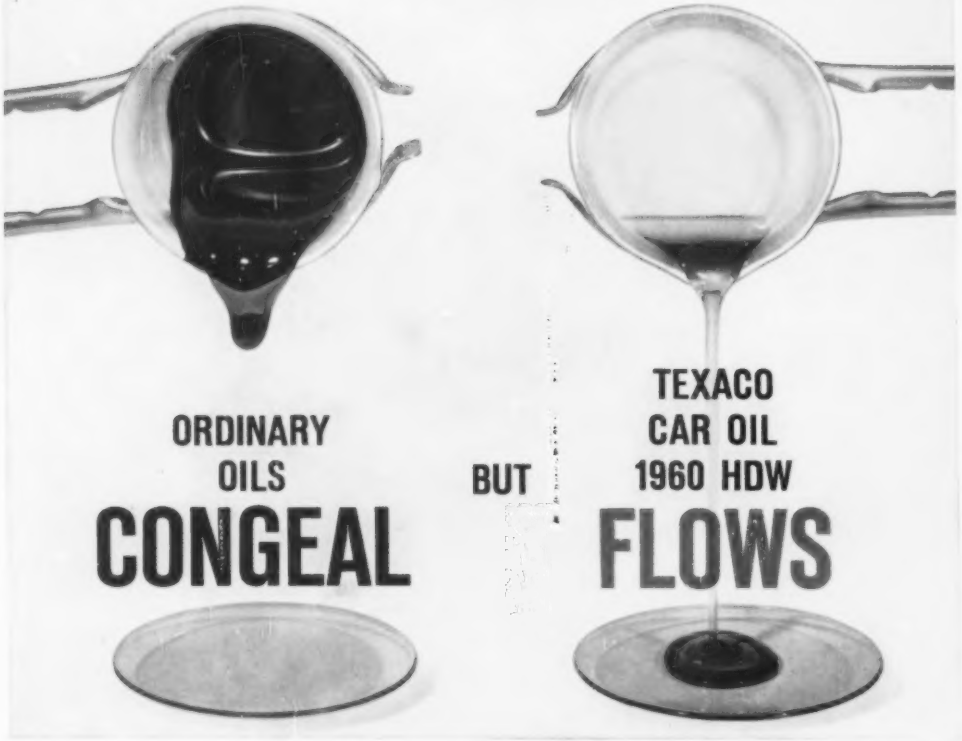

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